

FIELD EXPERIENCE AND SOLUTION OF A FRACTIONAL FREQUENCY VIBRATION PROBLEM ON A HIGH SPEED CENTRIFUGAL COMPRESSOR

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ABSTRACT

This paper describes how a serious field vibration problem on a high speed centrifugal compressor using tilting pad bearings was identified, following the introduction in the plant of simple methods of vibration recording and analysis, as being due to fractional frequency whirl at around 40% of the running speed. A clear correlation was discovered between the vibration behavior and the bearing clearance. This correlation is shown to match closely the predictions of a theoretical study of the effect of bearing geometry (clearance and preload) on the stability of the particular rotor/bearing system, when a commonly used empirical criterion for stability is employed, namely that the logarithmic decrement of the system should be at least 0.25 for instability to be avoided.

The study shows that adequate stability could only be achieved in this particular system for bearings lying within a limited range of clearance and preload. Tight clearance, high preload bearings or loose clearance, negative preload bearings both give instability, and variations in bearing dimensions between nominally identical units are shown to be capable of moving the system from the stable to the unstable zone. The vibration problems have now been overcome by proper pad selection aimed at achieving an optimum bearing clearance and preload of about 7 mils and 0.15, respectively.

The marginal stability in this machine is associated with the fact that the shaft first critical speed is about 44% of the full running speed, while machines of the same type, but with stiffer shafts giving a critical speed of about 55% of the running speed, have been free of instability problems. To provide a long term solution to the problem, and to cater for some increase of the maximum running speed in the future, stiffer rotors are being manufactured for the machines to raise the

rigid bearing first critical speed to 57% of the current maximum running speed.

In future purchase specifications for high pressure compressors a stability analysis will be requested for machines having a rigid bearing critical less than 60% of the maximum running speed, which will be required to demonstrate a logarithmic decrement of not less than 0.4 over the full possible range of bearing geometries, allowing for shaft, housing and pad tolerances, and wear. A main recommendation of the paper is that a refined rule of this type should be developed and incorporated in API 617.

INTRODUCTION

The equipment discussed in this paper is an eight-stage high pressure centrifugal compressor, in natural gas compression duty, on unmanned plants in Venezuela's Lake Maracaibo oil field. The layout and typical operating conditions are shown in Figure 1. Ten plants of broadly the same design are now installed, the five original units being in service since 1971/1972.

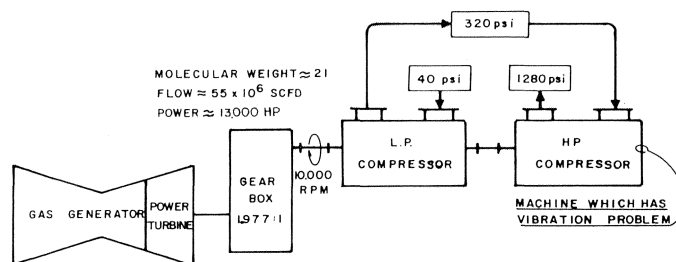


Figure 1. Machinery Schematic and Typical Operating Conditions.

The original high pressure (HP) compressor design had to be modified to raise the rotor critical speed before the units could be run at all. Subsequently, reliability of the HP compressors has been poor due to lateral vibration problems which have necessitated frequent bearing and seal replacement in the field, and return of the compressors to the shops for overhaul about once every eight months, for the original five units. One unit, identified as TJ-1, could not be run satisfactorily between March 1977 and April 1978. Serious production losses and repair and maintenance costs have resulted.

The maintenance department has sought to control the problem principally by changes to the bearing geometry and the oil seal clearances, since these changes were discovered to influence the vibration behavior, and were also relatively easy to introduce. However, the search for a solution was hampered

by inconsistencies in the response of the vibration level to the changes introduced, and by the absence of any information on the frequency content of the vibration.

During the experience with TJ-1 in 1978, vibration spectra were obtained using the services and analysis equipment of external consultants, which showed clearly the appearance of high amplitudes of vibration at about 50% of the running speed. This provided a positive identification of rotor instability as a major factor in the problems, and led to the introduction of vibration spectrum monitoring as a standard troubleshooting tool and to the commissioning of a stability analysis of the compressor.

Correlation of the collected field vibration spectra with dimensional variations of the bearings and seals and with the results of the stability analysis eventually identified the nature of the problem and appropriate solutions are described herein.

FIELD OPERATING EXPERIENCE

A survey of field operating experience was completed early in 1980, using the records of the machine build and vibration spectra from the previous 30 months of operation (that is, from the commencement of vibration spectra recording). The main factor which emerged from the survey was a marked influence of bearing geometry on vibration behavior. The effect of the clearance in the floating oil seal rings on the low pressure side of each seal was also investigated since this was believed to be a potentially influential factor on rotor stability. The effect of running condition could not be explored as the available records all related to a narrow range of compressor speed (close to 10,000 RPM) and discharge pressure (about 1250 psi).

To quantify the results of the survey, Figure 2 summarises the vibration behavior and build data collected for the various units at various times. Each plotted result on Figure 2 records the following data:

1. Plant location and date of vibration check.
2. Which of two possible types of pads were installed.
3. Assembled bearing diametral clearance in mils for each of the two bearings of the machine (based on a dimensional survey at last machine build of the shaft and housing diameters and pad thickness).

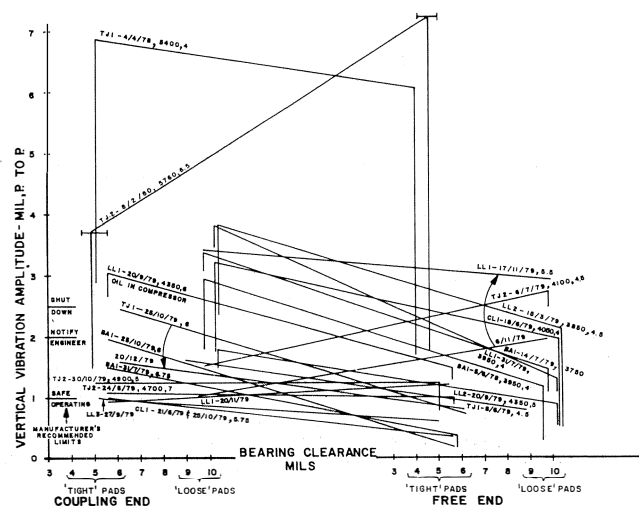


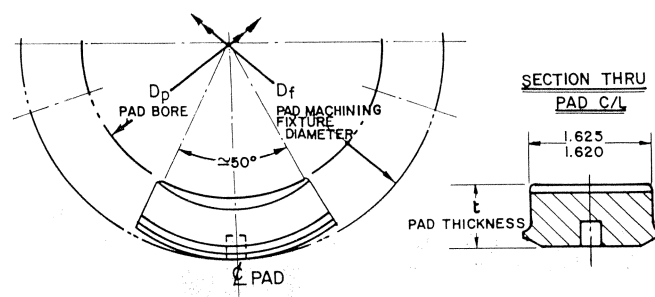
Figure 2. Summary of Field Vibration Experience as a Function of Bearing Clearance.

4. Average diametral clearance of the low pressure (LP) oil seal rings at each end of the machine, in mils.
5. Total vibration amplitude at each bearing (all frequencies), in mils peak-to-peak.
6. The frequency of the predominant fractional vibration for the rotor, if any, in CPM.
7. The total amplitude of the fractional frequency vibration at each bearing (in mils peak-to-peak).

The vibration performance was derived from proximity probes, located immediately adjacent to each bearing and viewing the shaft vertical displacement. The frequency content of the vibration signal from each probe was obtained with the aid of a tuneable filter device (Bently Nevada TK-8). The last two items, 6 and 7 above, were used to quantify the degree of instability shown by the machine, if any.

Figure 2 plots the total vibration amplitude (data item 5) versus bearing clearance (data item 3), both for the coupling (discharge) end bearing and the free (suction) end bearing, separately. The two results are linked by a near-horizontal line representing the rotor, against which is noted data items 1, 6, and 4, in that order.

Pad type (data item 2) may be inferred from the bearing clearance values. Only two types, as shown in Figure 3, have been used and only five-pad bearings of the load-between-pad type have been used in the field and considered in this study. The tight and loose clearance types give assembled clearance values (depending on component tolerances) in the ranges 4 to 6 mils and 9 to 11 mils, respectively. Consequently the group of 'rotor lines' displaced to the left of Figure 2 are for machines using pads of the tight clearance type, and the group of lines displaced to the right are for machines using pads of the loose clearance type. The amplitudes of the fractional frequency vibration (data item 7) in each bearing (if any) are shown by the length of the vertical 'legs' dropped from each end of the 'rotor lines'.



PAD TYPE	PAD BORE, D _p	PAD THICKNESS, t	FIXTURE DIA-D _f
TIGHT CLEARANCE	4.006/4.007	.8090/.8095	5.625
LOOSE CLEARANCE	4.008/4.009	.8070/.8075	5.623

NOTE: BEARINGS OF THIS STUDY ARE OF 5-PAD, LOAD-BETWEEN-PADS TYPE

Figure 3. Dimensions of the Two Pad Types Used.

Figure 2 reveals the following:

1. Vibration problems occur almost exclusively on the five original units which have rotors of lower stiffness. It can be seen that almost all the results on Figure 2 are for the five original plants: BA-1, LL-1, LL-2, TJ-1, and TJ-2. This reflects the fact that far more vibration

problems, and hence the need to record vibration behavior, have been seen with the original units. The main difference from a rotor dynamics point of view between the original and subsequent units lies in the first critical speed values for the various units, which are shown in Table 1. Since increasing rotor stiffness and hence first critical speed is known to increase rotor stability [1, 2], there is a strong implication that the vibration problems with the five original units were due to a fractional frequency vibration associated with reduced stability of their more flexible rotor design.

2. For rotors carried in loose clearance bearings rotor vibration is generally high, with much evidence of fractional frequency vibration (instability). This can be seen from the 'rotor lines' displaced to the right in Figure 2, which represent rotors in the loose clearance bearings and which, in many cases, lie above the manufacturer's recommended limits and have vertical 'legs' indicating more or less high amplitudes of fractional frequency vibration.
3. For rotors carried in tight clearance bearings rotor vibration is generally acceptable, with minimal evidence of fractional frequency vibration. This is shown by the rotor lines displaced to the left in Figure 2, which show total vibration generally within the manufacturer's limits and little or no vertical 'legs'.

Actually only four results show unacceptable vibration, and two of these can be discounted. Firstly, the high vibration at TJ-1 on 25/10/79 contained a significant X2 component indicative of misalignment; a further check on 20/12/79 on the same machine following realignment showed a big reduction of the high vibration at the coupling end. Secondly, the high vibration at LL-1 on 20/09/79 was associated with a severe lube oil leak into the compressor past defective oil seals, which is believed to have contributed to the vibration. The remaining two high vibration results are completely untypical and are accounted for below.

4. The unique pair of results showing extremely high vibration and instability in the tight clearance bearings were associated with two builds which, due to dimen-

sional variations of the bearing components, had uniquely low bearing clearances. No other cases of such low clearances, or such severe vibration, have occurred.

5. The frequencies of the fractional frequency vibrations of the five Phase 1 units, when present, are approximately equal to the calculated rigid bearing first critical speed of the Phase 1 rotors, again implying excitation of the rotor first critical by some instability mechanism. This is shown more clearly in Figure 4, in which the dominant natural frequency is plotted against the average bearing clearance and compared with various theoretical values. Further evidence regarding the rotor critical speed is shown in Figure 5, which shows rotor vibration versus speed during run-ups of Phase 1 units on two occasions. The maximum response values from Figure 5 have been added to Figure 4 also. It may be noted that the frequency of the severe fractional frequency vibrations in tight clearance bearings lie significantly above the calculated rigid bearing first critical,

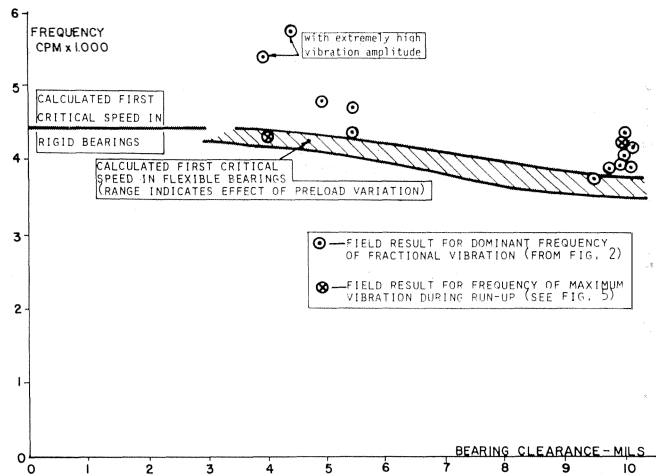


Figure 4. Calculated and Field Results for Critical Speeds.

TABLE 1. CRITICAL SPEEDS AND VIBRATION CHARACTERISTICS OF VARIOUS ROTOR DESIGNS.

Rotor Identification	Design Features	Critical Speed in Rigid Bearings RPM	Behaviour In Service
Original		3905	Could not be run at full speed due to severe vibration.
Phase 1 (Subject of this paper)	4.375 in. shaft of reduced span (55 in.); $\delta=0.47$ with optimum bearing geometry.	4463	Instability and vibration problems except with near optimum bearing geometry.
Phase 1.5	Increased shaft diameter	5460	Generally good and without evidence of instability.
Phase 2.0	Increased shaft diameter, plus lighter impellers.	5597	
New design to replace Phase 1 — in manufacture.	5 in. shaft of reduced span (53.5 in.); $\delta=1.038$ with optimum bearing geometry	5680	

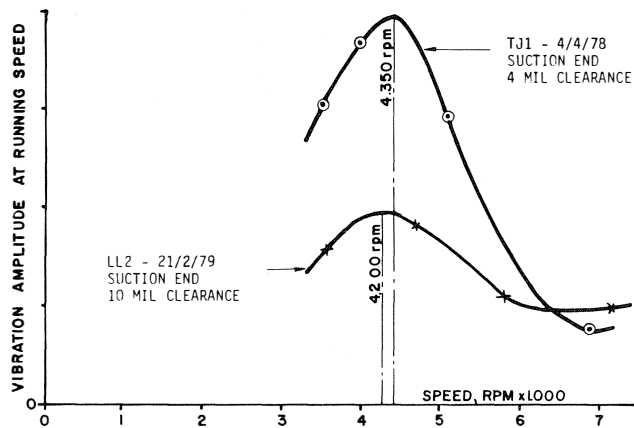


Figure 5. Vibration vs Running Speed during Machine Run-up.

and in the case of TJ-1 on 4/4/78, above the observed maximum response frequency during run-up as well. The reason for this is not clear, although it could be due to the effective support points of the rotor moving inward as a result of the very high amplitudes observed in these cases, and the consequent rubbing or contact between the rotor and internal parts, such as oil seals or labyrinths.

- The limited available field evidence suggests that the clearance in the LP oil seals does not have any significant effect on the dynamics of the rotors supported in tight clearance bearings (Figure 6), but could be provoking destabilization of the rotors in loose clearance bearings when reduced below 5.5 mils (Figure 7).

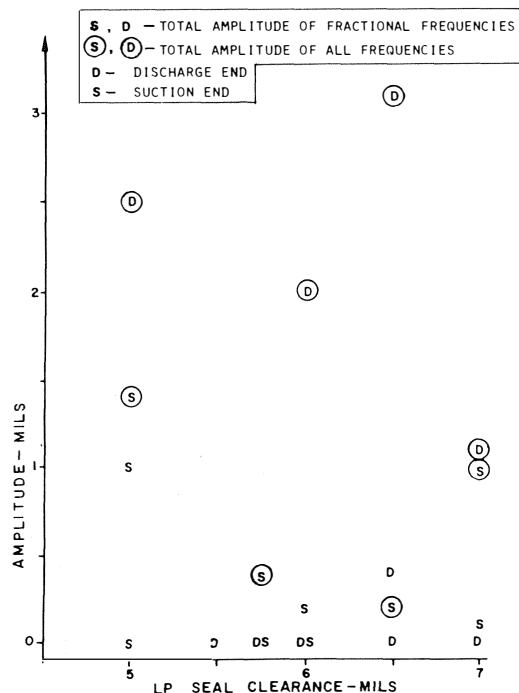


Figure 6. Vibration vs LP Seal Clearance, for Rotors in Bearings with Five to Six Mils Clearance.

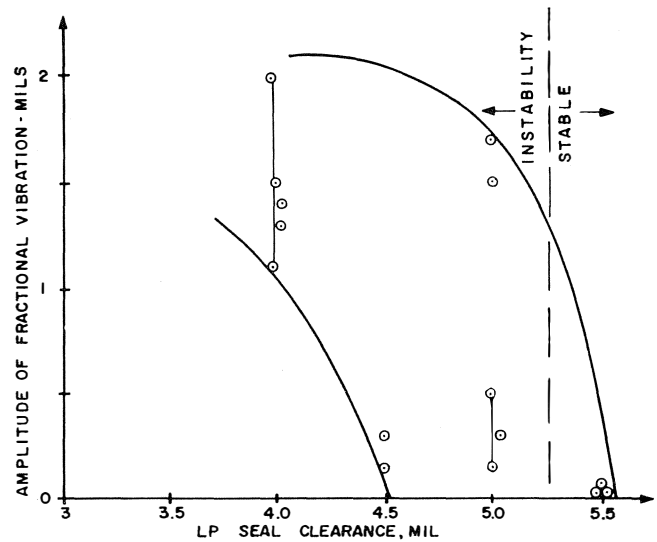


Figure 7. Amplitude of Fractional Frequency Vibration (for Rotors in Loose Clearance Bearings) vs LP Seal Clearance.

This evidence from practical operating experience has been supported and, to a large extent, explained by a computer study of the rotor stability as a function of bearing geometry, which was carried out by a major research and engineering organization and reported in mid-1979.

STABILITY ANALYSIS AND THEORETICAL INFLUENCE OF BEARING GEOMETRY

The results of the study by the engineering organization are shown in Figure 8. They show the effect of assembled

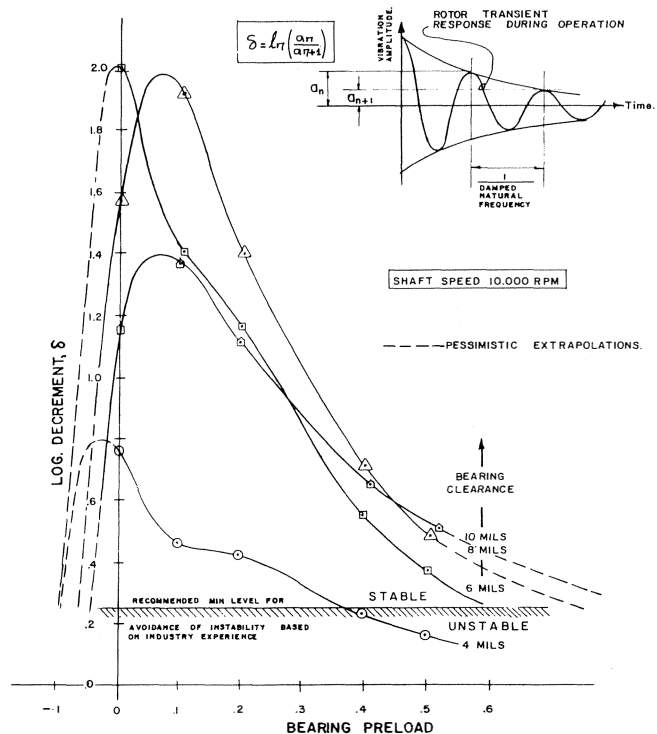


Figure 8. Logarithmic Decrement (a Measure of Rotor Stability) as a Function of Bearing Clearance and Preload.

bearing clearance and preload on the effective damping of the rotor when executing damped, unforced vibrations at its first critical speed. The measure used for the damping is the logarithmic decrement, δ , which shows the rate of decay of the vibrations, as indicated on Figure 8. Based on practical experience, a widely used criterion for adequate stability in centrifugal compressors is that the log decrement should exceed 0.25 [1].

Using this criterion, Figure 8 indicates that there should be limits to the bearing geometry beyond which instability is possible. For example, a bearing with 4 mils clearance should not have a preload in excess of 0.375, and one with 6 mils clearance should not have a preload greater than 0.6.

Figure 8 also suggests the existence of a lower limit to preload, due to the log decrement versus preload curves peaking and then falling as the preload reduces. Unfortunately the study was not continued to negative preload, but reference to other studies [2] suggests that the curves will drop appreciably more sharply with reducing preload than with increasing preload. Furthermore, the general experience that use of negative preload is to be avoided also suggests the existence of a limit at quite a small negative preload. In Figure 8 the dashed portions of the curves at negative preloads are conservative possible extrapolations, and Figure 9 is a transformation to show more clearly the resulting implied limits to bearing geometry.

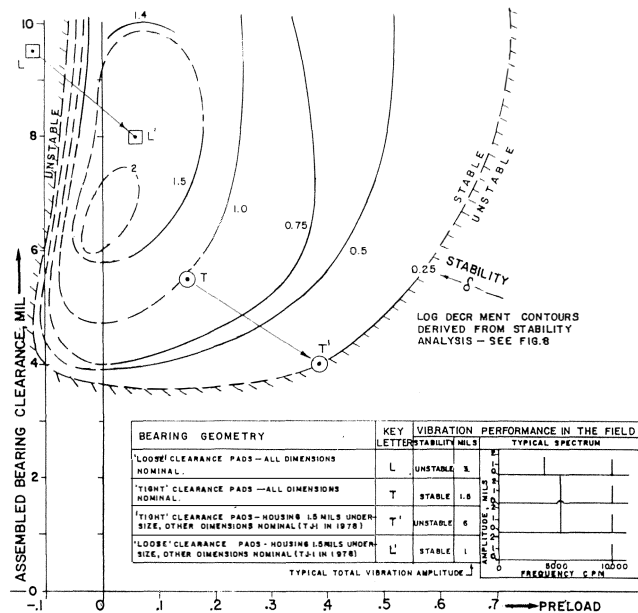


Figure 9. Stability vs Clearance and Preload — Comparison of Theoretical and Field Results.

GOOD AGREEMENT OF THE THEORETICAL PREDICTIONS WITH THE FIELD VIBRATION EXPERIENCE

By locating on Figure 9 various bearing geometries actually used in the field, it can be seen that the theoretical results predict very well the observed field vibration experience. Point L represents the bearing geometry with loose clearance pads and shows that it has a negative preload of -0.118 and is outside the area of predicted stability, thus confirming the practical experience.

Point T represents the bearing geometry with tight clearance pads, and predicts the stable operation generally observed in practice with the tight clearance bearings. Point T' predicts the effect on the geometry, with tight clearance pads, of the housing being undersize by 1.5 mils, and predicts the possibility of unstable operation with this modified geometry. This was in fact the geometry which gave the very severe unstable operation of TJ-1 in 1978. It may also be noted that the solution eventually adopted in that case — to revert to loose clearance bearings — gave operation which was uncharacteristically smooth for loose clearance bearings. The explanation is indicated by point L' in Figure 8, which shows the bearing geometry, with loose clearance bearings in an undersize housing, moved close to the most stable operating area, with a log decrement in excess of 1.5. The stability analysis thus appears to accurately define the field problem and can therefore be used as a basis for optimizing bearing geometry.

PROBLEM SOLUTION BY BETTER CONTROL OF BEARING GEOMETRY

Solution to the problem can be seen to lie in the selection of pad dimensions, such that the resulting bearing geometry, with any combination of shaft and housing diameters likely to be met in service, will lie comfortably within the stable zone. In order to assess the effect of pad dimensions and component dimensional tolerances on stability, it is more convenient to redraw Figure 9 using different axes which are simple functions of the four machined dimensions needed to define bearing geometry, namely:

- D_h = Housing diameter,
- D_s = Shaft diameter,
- D_p = Pad bearing surface diameter,
- D_f = Diameter of fixture against which the pad pivots seat while the bearing surface is being machined.

Figure 10 shows how the bearing clearances and preload relate to these four machining dimensions, and Figure 11

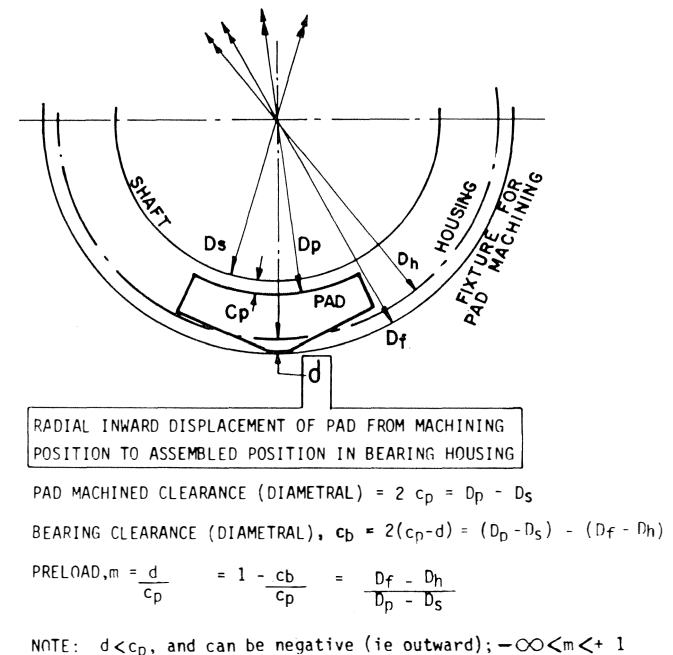


Figure 10. Clearance and Preload vs Machining Dimensions.

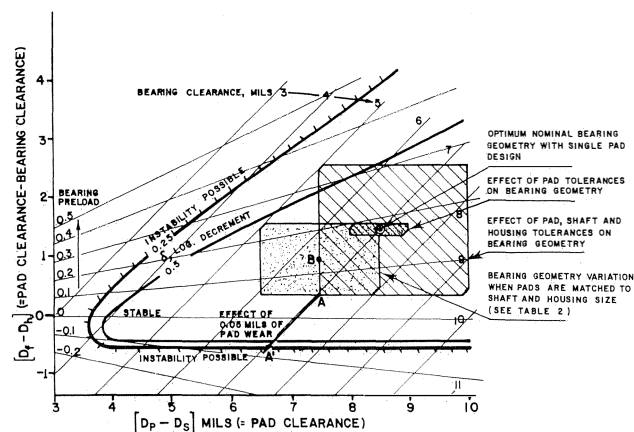


Figure 11. Stability Map for Selection of Optimum Bearing Geometry Allowing for Pad, Shaft and Housing Tolerances, and Possible Effect of Pad Wear.

shows the stability map redrawn with axes of $D_f - D_h$ and $D_p - D_s$ which readily permit the influence of machining tolerances to be seen. $D_p - D_s$ is simply the pad diametral clearance. $D_f - D_h$ may be interpreted as the pad clearance minus the bearing clearance, or the amount, e , that the pads must be moved in from the machining position to the installed position.

Included on Figure 11 are the log decrement contours, $S = 0.5$ and $S = 0.25$, and bearing clearance and preload lines. The $S = 0.25$ contour is assumed to define the stable zone based on the good apparent correlation of this criterion with the previous field experience.

The values of the four machined dimensions are as follows. The spread of housing diameters, D_h , as revealed by a survey of the Phase 1 machines actually in service at the end of 1979, was 5.623 in. to 5.625 in. and journal diameters, D_s , currently lie between 4.0005 in. and 3.999 in. Existing pad dimensions are as shown in Figure 3. The tolerance on the pad machined diameter was ± 0.0005 in., and although not stated or available from the manufacturer, it was assumed that the fixture diameter, D_f , was held to tight tolerances of ± 0.0001 in.

For housing and shaft diameters having the nominal values of 5.624 in. and 4.000 in., respectively, and pad dimensions of, for example, $D_f = 5.6255 \pm 0.0001$ in. and $D_p = 4.0085 \pm 0.0005$ in., the bearing geometry would lie within the small cross hatched area on Figure 11. This area thus indicates the effect on the bearing geometry of pad dimensional variation within tolerances. The depth of the area reflects the tolerance on D_f , the width reflects tolerance on D_p and the missing corners reflect tolerance on pad thickness (of ± 0.00025 in. for the pads used here).

The effect of dimensional variation of the housing and shaft diameters is to move the area representing the bearing geometry by 1 mil upwards/downwards for 1 mil of housing undersize/oversize and by 1 mil rightwards/leftwards for 1 mil of shaft undersize/oversize, respectively. Consequently, for the existing spread of housing and shaft diameters ($D_h = 5.623$ in. to 5.625 in., and $D_s = 3.999$ in. to 4.0005 in.), the bearing geometry with the above mentioned pad size could lie within the larger cross-hatched area on Figure 11. It can now be seen that this pad size would be a good choice for the prevailing spread of shaft and housing sizes, in that the resulting area representing the bearing geometry is completely contained within the stable zone.

This choice also provides some allowance for wear, which sometimes is seen as polishing of the central region of the pads and which is believed to result from occasional incidents of high vibration due to inadvertent operation in surge or other abnormal condition. Wear of this type, if assumed to be distributed so as to maintain a pad surface having a circular arc, but of slightly reduced radius, would move the geometry in the direction of the arrow shown in Figure 11; i.e., it would reduce preload with hardly any effect on clearance. This is because, from the geometry of circular arcs (Figure 12), an increase in radial bearing clearance of 0.05 mil, say due to wear, at the pad center tapering to zero at the ends, would imply a reduction of effective pad machined bore, D_p , of 1 mil, which would move the geometry from A to A' (Figure 11). This slight pad wear could therefore, with the most adverse tolerance stack-up, cause instability to occur.

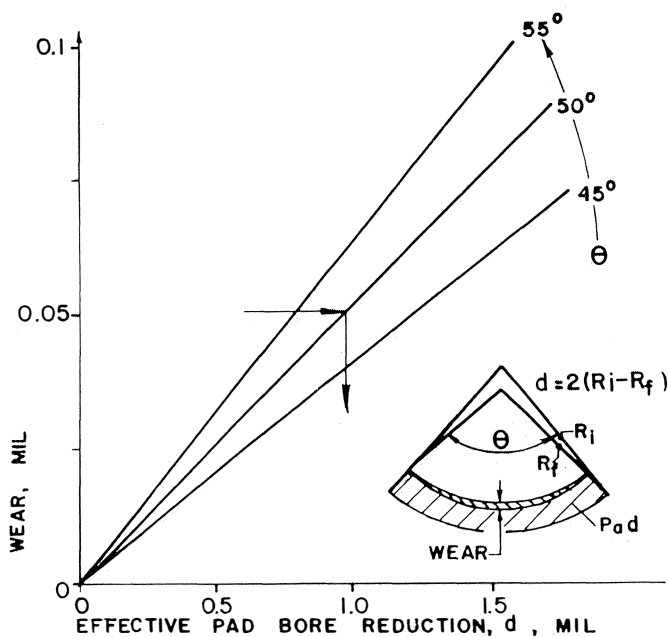


Figure 12. Effect of Wear on Pad Geometry.

The significant effect of pad wear should be noted and implies that it is desirable to choose a new pad geometry which initially provides more preload than optimum. Although Figure 11 suggests that an acceptable solution can be found with an optimum pad selection, it is clear that the margin of safety against instability, with most adverse tolerance stack-up and any pad wear, is very small, especially since the instability boundaries can only be considered to be approximately defined. As a more reliable temporary solution to the problem, it was decided to select and purchase pads in six sizes, each matched to a particular range of housing and shaft diameters (Table 2) to ensure that the bearing geometry could not diverge significantly from an optimum target geometry defined by point B (Figure 11). By carefully measuring the shaft and housing diameters prior to shop rebuild and by selecting an appropriate pad size from Table 2, the bearing geometry can be kept within the limits indicated by the dotted area on Figure 11, thereby appreciably increasing the margin against instability.

Since the introduction in the spring of 1980 of better control of the bearing geometry in line with the conclusions of

TABLE 2. PAD SIZES MATCHED TO SHAFT AND HOUSING DIAMETER TO GIVE BETTER CONTROL OF BEARING GEOMETRY.

Shaft Diameter Range (Inches)	Housing Diameter Range (Inches)	Pad Dimensions for Target Geometry of: Bearing Clearance = 6.5 Mils, Preload = 0.13 (i.e., Point B on Fig. 11)	
		Pad Bore, D_p (in.)	Pad thickness, t (in.)
4.0005-3.9995	5.6255-5.6245	4.007-4.008	.8090- .8095
"	5.6245-5.6235	"	.8085- .8090
"	5.6235-5.6225	"	.8080- .8085
3.9995-3.9985	5.6255-5.6245	4.006-4.007	.8095- .8100
"	5.6245-5.6235	"	.8090- .8095
"	5.6235-5.6225	"	.8085- .8090

this study, no further problems of instability and associated high vibration have been experienced.

LONG TERM SOLUTION BY RAISING SHAFT STIFFNESS AND HENCE ROTOR CRITICAL SPEED

As a long term solution, it was considered desirable to further increase rotor stability by raising the shaft stiffness and, hence, critical speed. The aim was to avoid the need for stocking and selecting six different pad sizes and to provide additional margin against the anticipated need eventually to raise the compressor maximum speed from the present value of 10300 RPM to 10870 RPM to maintain delivery pressure when the gas molecular weight drops in the future as a result of the removal of the richer fractions for LPG manufacture.

The compressor manufacturer has developed a modified rotor and bearing housing design which allows the shaft diameter to be increased by 14.3% and the bearing span to be reduced by 3% without adversely affecting the aerodynamic performance. These changes increase the rigid bearing critical speed from the present 4460 RPM for the troublesome Phase 1 units to 5680 RPM, which is higher than that of the Phase 1.5 and Phase 2 units (Table 1), which have not suffered instability problems.

The manufacturer has performed a stability analysis on the new design and has reported an adequate margin of stability even at the future maximum speed, and based on this, manufacture is now well underway.

In the meantime, a full lateral vibration analysis by the manufacturer's consultant has been commissioned, the results of which are awaited. They will be used to prepare a stability map of the type shown in Figure 11 to check the stability of the revised rotor at the future maximum speed, the suitability of the proposed pad type, and the margins against instability with the most adverse tolerance stack-up and pad wear.

In the future this type of information will be sought from manufacturers when purchasing new high pressure compressors, especially when the first bending critical in rigid bearings lies below 60% of the maximum running speed. By this means it is expected to be able to avoid a repetition of the major problem and expense experienced with the subject units. It is also suggested strongly that the API Specification 617 should be modified to incorporate a requirement for manufacturers to provide this type of information.

CONCLUSIONS

1. Tilting pad bearings cannot be relied on to prevent fractional frequency whirl of flexible rotors and, as with

plain bearings, it appears that the shaft critical speed must be kept above about half the running speed to prevent instability. It is recommended that an addition be made to API Specification 617 requiring that a stability analysis be carried out by the manufacturer on all high pressure compressor rotors having a first critical speed in rigid bearings less than 60% of the maximum running speed. This testing must demonstrate a log decrement in excess of 0.4 at all combinations of bearing dimensions within tolerance and allowing for a reduction of preload of 0.1 due to pad wear. The results should be presented in the form of a stability map of the type shown in Figure 11.

2. In the absence of well established guidelines for avoiding or eliminating instability problems in centrifugal compressors, the following recommendations for users are offered:
 - 2.1 Favor designs with first critical speeds, in rigid bearings, of 60% or more of the maximum running speed.
 - 2.2 For all designs ideally, but certainly if the first critical speed in rigid bearings is below 60% of the running speed, obtain a stability analysis for the machine, including determination of the effect on stability of bearing clearances and preload and prepare a stability map of the type shown in Figure 11. Use this to select an optimum bearing geometry giving a log decrement for the rotor first critical of no less than 0.4 after allowing for the effect of realistic shaft, housing and pad tolerances, and slight pad wear.
 - 2.3 Where stability problems are suspected in the field and in the absence of a stability analysis or other guidance, experiment with different bearing clearances by pad changes, but also aim for a preload of 0.15 and ensure that the preload stays within the range 0 to 0.3 even under the most adverse tolerance stack-up.
 - 2.4 If a machine is already built and the stability is marginal even with an optimum pad design, the stability will be very sensitive to component tolerances and it will be worth considering the possibility of:
 - a) Tightening the tolerances on shaft, housing and pad diameters.
 - b) Using several pad sizes and matching them to each shaft and housing diameter combination as measured.

- c) Modifying labyrinths to reduce aerodynamic excitation [2].
- 2.5 In the worst case, that adequate stability cannot be achieved with optimized bearing geometry, the remaining options capable of achieving a substantial increase of stability at the cost of major redesign are:
 - a) Raise the first critical speed by stiffening the rotor.
 - b) Design in squeeze film damping of the bearings.
- 3. Some form of spectrum analysis is an essential tool for investigating field vibration problems. Instability will be shown by an appreciable component of vibration at the first natural critical frequency of the rotor. This will normally be somewhat below the rigid bearing critical speed, but higher values may be seen if the vibration amplitude is very high. Values up to 1.3 times the rigid bearing critical and 57% of the running speed were

seen in the present study. It cannot therefore be assumed that instability is only evidenced by vibration at 50% or less of the running speed.

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